# DIRECT-EXPANSION AIR-CONDITIONING SYSTEM PERFORMANCE IN LOW HUMIDITY APPLICATIONS: A CASE STUDY

MUKESH K. KHATTAR, P.E.
Senior Systems Engineer
Florida Solar Energy Center
Cape Canaveral, Florida 32920

DENNIS KEEBAUGH, P.E.
Senior Research Engineer
Shenandoah Solar Center
Georgia Power Company
Shenandoah, Georgia 30265

## ABSTRACT

This paper discusses the performance of an HVAC system to maintain temperature and humidity within a candy storage warehouse. The HVAC system consists of a typical direct-expansion cooling and dehumidification coil and electric resistance reheaters. Historic energy-use data and field-measured performance data are analyzed. The importance of certain design and operating parameters such as indoor airflow quantity, cooling coil apparatus dew-point temperature and the air-handling unit's fan operating strategy on the dehumidification performance of the HVAC system are discussed and analyzed. It also offers recommendations for design and operation of such systems.

## INTRODUCTION

Storage of candy canes requires low humidity to retain texture and product quality. High humidity bleeds color from stripes and makes them sticky. One of the largest candy manufacturer maintains 38 to 42 percent relative humidity and 76 to 80 F temperature in their warehouses. The performance of the air-conditioning system is one of their 45,000 ft$^2$ warehouse in Albany, Georgia is the subject of this study. This warehouse is used for seasonal storage during summer; the product is brought in during April/May and removed for shipment in October/November. The north, east and south exterior walls are constructed with brick without any insulation. The west wall forms a partition with an adjoining unconditioned warehouse. The flat gravel roof is exposed to sun. The 16' ceiling is insulated with two inch spray foam. Entrance to the warehouse is through sealed and insulated doors located on the west partition wall.

The air-conditioning system on this leased warehouse was designed and installed by a local air-conditioning contractor. The system consists of two identical rooftop units, each consisting of one air-cooled condensing unit of 30-ton nominal capacity, one air-handling unit with 6-row deep direct-expansion cooling coil, and 80 kW electric reheat in two steps. One unit is located in the middle of the north half of the warehouse, and the other unit is located in the middle of the south half. The warehouse does not have any internal partition, and both units are serving the single zone. Each unit is controlled by an independent set of electrical controls. The system controls are typical: the compressor is controlled by the higher of humidistat or thermostat signal; and the reheat is controlled by a two-stage thermostat. The indoor-air fan was originally set to operate continuously regardless of the compressor cycling. In this mode of operation, some of the condensed moisture re-evaporates when compressor cycles off. This mode of operation has been changed so that the fan cycles on and off with compressor cycling.

## PERFORMANCE MEASUREMENTS

The following measurements were taken to characterize the system performance.

### Electrical Energy-Use Profile

Four utility load research meters were installed to record electrical demand profile for every fifteen minutes on the following circuits: combined energy use by the condensing and air-handling units for each of the north and south units, and electric reheat by each of the north and south units. This provided a diurnal system load profile as well as total energy use by each subsystem for October and November of 1986. In addition, total electrical energy use by the air-conditioning system alone for the past four years is available from the utility billing data.



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## Condensate Measurement

The primary purpose of the air-conditioning system is to remove moisture. Excess sensible cooling which thermodynamically accompanies dehumidification is removed by air-reheat. The moisture removed by the system collects in the condensate drain pan of the air-handling unit before it is drained out. Separate flow meters were installed in the condensate drain line of each of the two air-handling units to measure the condensate flow. The flow meter provided resolution of 200 pulses per gallon, and was read manually every day.

## Air-side Performance

The following parameters were measured on a one-time basis to characterize the air-side system performance. The measured data and calculated system performance is shown in Figure 1.

Air Flow Rate. The air-flow rate was measured with a rotating vane anemometer by scanning cross the return air opening. Several readings were carefully taken to reduce error that may be introduced due to scanning rate and pattern. The measurements were repeated with a hot wire anemometer. The readings with different sensors were consistent and within five percent of the average value.

Return Air Conditions. The dry-bulb temperature was measured at the return air (RA) opening with a platinum RTD sensor. The relative humidity was measured with a thin-film capacitance type sensor. The sensors were scanned across the RA opening during measurement. Measurements were also made with hand-held dry-bulb/wet-bulb thermometers.

Supply Air Conditions. The supply air (SA) temperature after the fan was measured by inserting a platinum RTD sensor in the SA duct. The sensor did not scan the duct. The SA temperature leaving the cooling coil was calculated by reducing the estimated temperature rise across the draw-thru fan using the measured fan power. The SA after the 6-row evaporator coil is assumed to be at 95% saturation level.

Fan Power. The power draw by the AHU fans was measured by a clip-on ammeter. A 50 percent fan efficiency was used in calculations of temperature rise across the draw-thru fan.

## ANALYSIS AND DISCUSSION

The cooling and dehumidification process with air reheat is shown on a psychrometric chart in Figure 2. The return air at state point 1 passes over the cooling coils and exits at state point 3. The SA at 3 provides sensible cooling proportional to the enthalpy difference between state points 3 and 6, and dehumidification proportional to the enthalpy difference between state points 1 and 6. Joining state points 1 and 3 by a line and extending the same to a point where it intersects the saturation line on the psychrometric chart (i.e.,state point 2) yields Apparatus Dew Point (ADP) temperature. The slope of this line uniquely determines the relationship between available sensible and latent cooling. A steeper slope indicates higher Dehumidification Fraction (DF), or lower Sensible Heat Ratio (SHR). By reducing the ADP temperature, within certain limits, it is possible to increase DF and reduce SHR. However, it can be seen from the psychrometric chart that by reducing ADP temperature below a certain level, no further improvement in SHR is available. This limit depends upon the entering air conditions or state point 1. The SHR versus ADP temperature of a cooling and dehumidification process for an entering air temperature of 78 F and a range of relative humidity values from 30 to 70 percent is shown in Figure 3. For a 40 percent relative humidity, the limiting SHR is about 0.71, and the corresponding ADP temperature is about 35 F. Although a lower temperature improves DF, it also reduces vapor compression system's energy efficiency. Further, ADP temperatures below 32 F runs the risk of condensate freeze-up and resulting damage to the cooling coil.

If the cooling and dehumidification load requires SHR below what a straight cooling and dehumidification process can provide, the system SHR must be modified by elevating supply air temperature by addition of heat, from state point 3 to 5 as shown in Figure 2. A part of the reheat, from state point 3 to 4, however, is provided by the fan in a draw-thru AHU. The amount of reheat can be estimated as follows:

$$\frac{RH}{TL} = \frac{SHR_a - SHR_1}{1 - SHR_a} \qquad (1)$$

or

$$\frac{RH}{AC} = \frac{SHR_a - SHR_1}{1 - SHR_1} \qquad (2)$$

where:

- RH = Reheat energy
- AC = Air-conditioning system capacity
- TL = Total cooling and dehumidification load
- $SHR_a$ = Sensible Heat Ratio of the straight air-conditioning process
- $SHR_l$ = Sensible Heat Ratio of the cooling and dehumidification load.

For example, if the cooling and dehumidification load $SHR_l$ is 0.55 and the air-conditioning system $SHR_a$ is 0.71, the air-conditioning system must be sized for 55 percent greater capacity than calculated from the total of sensible and latent cooling loads to overcome the added reheat. Further, 36 percent of the total energy used by the air-conditioning equipment can be termed as excess cooling which must be compensated by reheat. Therefore, not only is energy needed for reheat, but additional energy is needed for excess cooling to overcome the reheat. For this reason, even if the reheat energy is cheaply available, care must be taken in design and operation to avoid excess cooling energy requirement.

Estimating load SHR is difficult. Almost the entire latent load in this warehouse is due to infiltration. The candy boxes are brought in from other conditioned storage facilities and does not add any significant moisture load. There is no accurate method of calculating infiltration. Therefore, the traditional method of calculating loads will not yield accurate SHR. A novel approach from the measured and historical energy-use data is presented here to estimate the cooling and dehumidification load SHR.

The electrical energy-use by each of the condensing units and the reheaters as measured by the load profile record meters is given in Table 1. Also presented is the reheat ratio, a ratio of the electrical energy use by the reheat elements to that by the combination of the condensing and the air-handling units. Table 1 shows that the reheat ratio varied between 2.11 to 2.36.

The reheat ratio can be related to the air-conditioning system $SHR_a$ as follows:

$$RR = \frac{RH * EFF * COP}{AC}$$

$$= \frac{(SHR_a - SHR_l) * EFF * COP}{1 - SHR_l} \qquad (3)$$

or

$$SHRl = \frac{EFF * COP * SHR_a - RR}{EFF * COP - RR} \qquad (4)$$

where:

- RR = Reheat Ratio, a ratio of electrical energy use by elements to that by the condensing and air handling units
- COP = Air-conditioning System Coefficient of Performance
- EFF = Reheater efficiency

The electrical heater efficiency is 100 percent. However, the AC system COP will vary depending upon the outdoor air temperature. Based on manufacturers' data and site measurements, the system operating COP is estimated to vary between 2.9 to 3.1. The air-conditioning system $SHR_a$ also varies with ambient conditions. $SHR_a$ was measured at 0.88 for the south and 0.91 for the north unit. For a COP of 3.0 and $SHR_a$ of 0.88, the average $SHR_l$ is calculated as below:

$$SHR_l = \frac{1.0 * 3.0 * .88 - 2.2}{1.0 * 3.0 - 2.2}$$

$$= 0.55$$

The historical energy-use by the air-conditioning system alone for the past four years is shown in Table 2. Only combined energy use data for the two condensing, air-handling and reheat units are available. In order to calculate reheat ratio, it is assumed that the energy-use by the condensing and air-handling units is the same as measured from the load profile recorder for the month of October 86. The monthly average reheat ratio varies between 1.93 to 2.51. The data for the beginning and ending month is not included since the unit may not have operated for the entire billing period. Incidentally, it is instructive to note that the total energy-use decreases during peak summer months, primarily due to reduction in reheat energy requirement.

The monthly average reheat ratio of 1.93 to 2.51 from the historical yearly energy-use data is quite close to 2.13 to 2.36 as measured with load profile recorder for the months of October & November 1986. Therefore, based on the annual energy-use data, it is also reasonable to estimate the load SHR to be 0.55. This approach of estimating SHR is significantly more accurate than estimating air-infiltration rates and calculating SHR.

The review of the hourly load profile data for October '86 revealed the following: 1) both of the units operated continuously; 2) the south unit used about 70 percent of the energy used by the north unit. Hence it is postulated that the south unit operated at two-thirds partial load (the condensing unit has three steps of loading); 3) the reheat energy-use by the south unit is also proportionately lower (about 78 percent) than the north unit. 4) the reheat ratio, the ratio of reheat to air-conditioning energy consumption, for the two units is nearly the same; 5) the total energy use including both condensing unit and reheaters was 4,875 kWh per day.

During this period the indoor air fan operated continuously. This is not a good control strategy for humidity control. Continuous fan operation can cause large amounts of condensed moisture remaining on the cooling coil to re-evaporate after the compressor shuts off on call from the thermostat or the humidistat. Experimental data with a residential size air-conditioning unit at the Florida Solar Energy Center has shown that as much as 19 percent of the moisture condensed in the first 10 minutes of operation re-evaporated back into the space in the next 10 minute cycle when the fan continued to operate after the compressor was turned off (Khattar, 1987a).

The air-handling unit fan operation control was modified in both units for the month of November '86. The fan was cycled on and off along with compressor. The following observations are made from the load profile recorder data: 1)the north unit operated for 28 percent and the south unit 58 percent of the time. This is substantially lower than nearly 100% for both units as observed for the previous month under continuous fan operation; 2) the south unit continued to operate at two-third partial load; 3)the reheat energy-use makes it also proportionately lower (about 62%); 4)the reheat ratio for the north and south units is nearly the same, 2.13 versus 2.18; 5)the total energy-use is reduced to only 2,540 kWh per day compared to from 4,875 kWh per day in fan 'continuous' operation mode, providing an impressive 48 percent energy savings. Besides re-evaporation of the condensed moisture into the space, fan cycling also reduces the fan-induced outdoor air infiltration into the space. However, some of the savings might be due to a milder weather in November than in October.

The importance of a lower system SHR in reducing reheat energy requirement and avoiding large increases in the equipment size is evident as shown in eqn. (1) and (2)earlier. The SHR of a cooling and dehumidification process can be characterized by a single operating parameter -- the apparatus dew-point (ADP) temperature -- which is the result of the interactions of several system design parameters and operating conditions. In a new system design, the various components of a system can be sized and controls can be set to yield desired SHR at the design as well as off-design operating conditions. In an existing system, however, all components have been pre-selected. Therefore, operating parameters such as airflow rate and ADP temperature can not be independently selected. Any variation in one will affect the other in a manner which is governed by the relative size of the system components. The following discussion underscores the importance of selecting a single parameter -- the airflow quantity -- of the existing DX system and how it affects the other operating parameters and system performance.

The simulation results of the direct-expansion air-conditioning system at the candy warehouse are presented in Table 3. The computer simulation performs iterations to match the capacity of the air-cooled condensing unit to that of the direct-expansion cooling coil for any given air flow rate and predicts sensible cooling capacity, total cooling capacity, compressor power draw, cooling coil saturated suction temperature, and air-side pressure drop through the coil. In these simulations, the conditions of air entering the cooling coil were maintained at 78 F temperature and 40 percent relative humidity. The outdoor air temperature entering condenser was 85F. The air-conditioning equipment manufacturer's computer program for equipment selection was used. The results were spot-checked with manufacturer's published performance data. The indoor air flow quantity was varied from 4500 to 14000 cfm in steps of 500. The results are discussed below.

<u>Air speed</u>. The air face velocity over the cooling coil is calculated by dividing the air-flow rate by the coil face area. Lower flow rates reduce heat removal rate from the coil and let its temperature to fall. Lower coil surface temperature reduces ADP temperature and removes more moisture. However, low velocities over the coil adversely affect the heat transfer coefficients. Some manufacturers recommend a lower limit of 300 fpm for the air speed. Higher air speed should also be avoided to reduce the risk of moisture carry over by the air.

<u>Saturated Suction Temperature (SST)</u>. The SST is the refrigerant saturation temperature corresponding to its pressure at the exit of the cooling coil. The SST at the suction part of the compressor will be lower than at the coil exit, typically by 2F. The SST decreases as the air flow decreases, and provides lower ADP temperature. However, lower SST also implies less total cooling capacity and lower system's energy efficiency. Although ADP temperature, or the average cooling coil surface temperature, is higher than the SST, SST below 32 F run the risk of condensing water to freeze on the coil. However, due to air moving over the coil and coil surface temperature being higher than the SST, some manufacturers permit SST up to 28 F without any need for defrosting. It is recommended to select SST to avoid frost build-up on the coil.

The SST varies with change in the ambient air temperature for an air-cooled direct-expansion unit. For this system, it decreased by 1.2 F for each 10 F drop in the outdoor air temperature. The outdoor air temperature may vary from 65 to 95 F during operation. Therefore, care should by taken in selection of SST at design conditions so that it is still within acceptable limit at part-load operation. This condensing unit is equipped with a compressor capacity control device actuated by the SST, which can be set to avoid SST below desired level.

<u>Dehumidification Capacity (DC)</u>. Dehumidification capacity is calculated by dividing the system's latent cooling capacity by the latent heat of evaporation of water. The dehumidification capacity and air speed are plotted against the cooling coil saturated suction temperature (SST) in Figure 4. The dehumidification capacity increases rapidly when the SST is reduced from 43 F, but the curve flattens out near lower SST of 35 F.

<u>Sensible Heat Ration (SHR)</u>. The SHR is derived from dividing the system's sensible cooling capacity by the total cooling capacity. As the SST reduces, the coil ADP temperature also reduces and SHR is reduced. The SHR is plotted against SST in Figure 5. The SHR drops rapidly when SST is reduced from 43 F, but the SHR curve flattens out at lower SST near 35 F.

<u>Fan Power</u>. The indoor air fan power is calculated to move the required amount of air through the air-handling unit and duct system. In a new duct design, the duct work can be reduced, and lower fan motor may be used for lower air flow. With an existing duct layout, still larger reduction in the fan power is possible because the power draw varies nearly with the third power of the air flow rate. The power draw in the two units was measured at 6.2 and 6.8 kW. Manufacturer's performance data was used for calculations of the fan power. The predicted pressure drop from the wet cooling coil was also obtained from the manufacturer's data. The pressure drop from the ducts, grills and filters was estimated to vary with air flow raised to the power of 1.8.

<u>System Energy Efficiency Ratio (EER)</u>. The system EER is the ratio of total cooling provided in Btu per Wh of electric consumption by the condensing unit and indoor air fan. The vapor compression system EER reduces at lower SST. But, the indoor fan power requirement also decreases at lower SST to handle less air. The drop in fan power is particularly significant in existing duct work, and the system EER actually improves at lower SST. The system EER for this application is plotted against SST in Figure 5.

<u>Cooling Energy Used</u>. The cooling energy requirement may be divided into two categories: useful and excess. All energy requirement is divided by the total dehumidification capacity to present data in terms of kWh energy required per pound by moisture removal. The useful and excess cooling energy requirement are calculated for a load SHR of 0.55.

The useful energy portion is used by the cooling and dehumidification system to remove moisture and provide necessary cooling. This corresponds to enthalpy difference between state points 1 and 5 in Figure 2. It is observed that the useful energy consumption increases slightly with reduction in SST.

The excess cooling energy is the amount of cooling that must be removed by reheat after the cooling and dehumidification process. It corresponds to the enthalpy difference between state points 4 and 5 in Figure 2. This cooling energy is required to remove the added reheat. This amount increases rapidly with increase in SST.

<u>Reheat Energy</u>. The reheat energy is required to add heat to the conditioned supply air. Electrical resistance heat is used in this application. The reheat energy is proportional to enthalpy difference between the state points 4 and 5 on the psychometric chart in Figure 2. The reheat energy is equal to the excess cooling energy multiplied by the COP of the cooling process less heat added by the draw-thru fan motor. The reheat energy also increases rapidly with increase in SST.

**Total Energy**. The total energy is the sum of the total cooling, reheat and fan energy required to remove each pound of moisture. The useful cooling energy, total cooling energy and total cooling and heating energy requirement are plotted against SST in Figure 6. The total energy requirement increases rapidly with increase in SST. For example an increase of SST from 35 to 37 F requires 23.2 percent more energy for moisture removal.

**Reheat Ratio**. Reheat Ratio is the ratio of electrical energy used for reheat divided by the total cooling energy. This ratio is helpful in estimating effective load SHR as discussed earlier.

## CONCLUSIONS

In the above analysis, it can be observed that there is quite a range of operating conditions where the system will work optimally. Although the energy requirement changes from .76 kWh to 1.1 kWh per pound of moisture removal when SST varies from 30 to 36 F, this change is relatively small compared to energy requirement at, say 40, when it needs about 2.2 kWh for every pound of moisture removal. The significance of lower SST is obvious.

In the field measurement of the system performance, it was found that two units were operating with large airflow quantities; 12,500 cfm for the south unit and 14,615 cfm for the north unit. Perhaps the systems were designed with typical comfort applications standards. Improvements in efficiency by reducing the air flow quantity does not add any cost. The system's operating conditions can be comfortably set for a total flow of 6000 cfm at corresponding SST of 33.8 F and obtain greater dehumidification efficiency as well as capacity. The selection of SST above 32 F also permits the SST to drop with a drop in outdoor ambient temperature and still avoid condensate freeze-up on the cooling coil. Further, since this condensing unit is equipped with compressor capacity modulation device, the SST can be set so that it does not fall below 32 F.

The change in fan control strategy from fan 'continuous' operation to fan 'cycling' has already resulted in 48 percent energy savings. Some of these savings may have been contributed by the weather. The change in fan control strategy also did not add any cost, but provided significant savings. This strategy is recommended where humidity control is important. Auxiliary fans may be used for air circulation, if needed.

The approach to estimate SHR from measured energy-use data as presented in this paper yields a far more accurate value than conventionally obtained from the calculation of latent load from estimation of air infiltration.

## FUTURE WORK AND ACKNOWLEDGEMENT.

This work was performed under a program designed to demonstrate a novel application of heat pipes to improve dehumidification performance of air-conditioning systems (Khattar, 1978b). Even at low SST of 33.8 F, excess cooling and reheating is required, as observed from Table 3. When a heat-pipe heat-recovery exchanger is placed between the warm air entering the cooling coil and the cold air leaving the cooling coil, free reheating as well as free pre-cooling is achieved (Khattar, 1986). This reduces and even eliminates the expense of excess cooling as well as reheating, and makes the system energy efficient. It also increases the dehumidification capacity of the system.

The heat pipes have been installed in one of the two units and the system is operational. Data is being collected for this summer season. The results will be published later.

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Table 1. Electric Load Profile Recorder Data

| Month | South Unit | | | | North Unit | | | | Total | | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| | Cooling kWh | Reheat kWh | Reheat Ratio | Run Time (%) | Cooling kWh | Reheat kWh | Reheat Ratio | Run Time (%) | Cooling kWh | Reheat kWh | Reheat Ratio |
| Oct'86 | 18,760 | 44,240 | 2.36 | 99.86 | 26,760 | 56,560 | 2.11 | 99.86 | 45,520 | 100,800 | 2.21 |
| Nov'86 | 10,240 | 21,800 | 2.13 | 58.04 | 7,680 | 16,760 | 2.18 | 27.69 | 17,920 | 38,560 | 2.15 |

Table 2. Energy Use by Cooling and Dehumification System

| | 1986 | | 1985 | | 1984 | | 1983 | |
|---|---|---|---|---|---|---|---|---|
| Month | Total kWh | Reheat Ratio | Total kWh | Reheat Ratio | Total kWh | Reheat Ratio | Total kWh | Reheat Ratio |
| JAN | 0 | | 0 | | 0 | | 0 | |
| FEB | 0 | | 0 | | 0 | | 0 | |
| MAR | 98680 | ** | 0 | | 0 | | 0 | |
| APR | 146000 | 2.21 | 43280 | ** | 0 | | 0 | |
| MAY | 147755 # | 2.25 | 159720 | 2.51 | 15480 | ** | 0 | |
| JUN | 124341 # | 1.73 | 152160 | 2.34 | 95840 | 1.11 | 115080 | 1.53 |
| JUL | 141207 # | 2.10 | 154400 | 2.39 | 111480 | 1.45 | 133280 | 1.93 |
| AUG | 125059 # | 1.75 | 91240 | 1.00 & | 143280 | 2.15 | 147920 | 2.25 |
| SEP | 130890 # | 1.88 | 140480 | 2.09 | 136960 | 2.01 | 138040 | 2.03 |
| OCT | 146320 * | 2.21 | 135600 | 1.98 | 119240 | 1.62 | 127720 | 1.81 |
| NOV | 56480 * | ** | 63560 | ** | 0 | ** | 0 | |
| DEC | 0 | | 0 | | 0 | | 0 | |
| Annual | 1116732 | 2.02 | 940440 | 2.26 | 622280 | 1.67 | 662040 | 1.91 |

All energy-use obtained from utility billing data except *
* Measured with Load Profile Recorder
\# Estimated, billing meter malfunctioned
& Only one of the two units operated, as observed from demand meter.

$$\text{Reheat Ratio} = \frac{\text{Electric Reheat Energy use, kWh}}{\text{Condensing Unit Energy Use, kWh}}$$

** System operated less than one month. Reheat ratio not meaningful.



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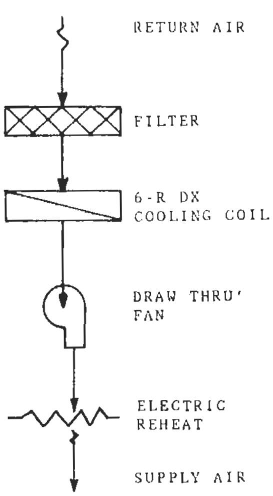
Table 3 Parametric Performance of a Direct-Expansion Air-Conditioning System with Electric Reheat for Humidity Control in a Candy Warehouse

Indoor Conditions: 78 F Dry-bulb Temperature, 40% Relative Humidity
Outdoor Air: 85 F; Load Sensible Heat Ratio: 0.55

| AIR-FLOW cfm | AIR SPEED fpm | SENS CAP Mbtu | TOTAL CAP Mbtu | DEH CAP lb/hr | SEN HEAT RATIO SHR | SAT SUCT TEMP F | COMP POWER kW | FAN POWER kW | SYS-TEM EER btu/Wh | COOLING ENERGY USEFUL kWh/lb | COOLING ENERGY EXCESS kWh/lb | REHEAT ENERGY kWh/lb | TOTAL ENERGY kWh/lb | REHEAT/COOLING RATIO kW/kW |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 4500 | 221 * | 208.4 | 294.5 | 80.1 | .708 | 29.0 * | 26.9 | 1.2 | 10.48 | .218 | .118 | .377 | .728 | 1.076 |
| 5000 | 245 * | 219.6 | 305.9 | 80.3 | .718 | 30.9 | 27.4 | 1.4 | 10.62 | .214 | .127 | .417 | .775 | 1.161 |
| 5500 | 270 * | 230.3 | 315.7 | 79.4 | .729 | 32.5 | 27.9 | 1.6 | 10.70 | .211 | .140 | .464 | .836 | 1.251 |
| 6000 | 294 * | 240.7 | 324.1 | 77.6 | .743 | 33.8 | 28.3 | 1.8 | 10.77 | .209 | .156 | .524 | .912 | 1.351 |
| 6500 | 319 | 250.7 | 331.6 | 75.3 | .756 | 35.0 | 28.7 | 2.1 | 10.77 | .207 | .175 | .591 | 1.000 | 1.444 |
| 7000 | 343 | 260.5 | 338.2 | 72.3 | .770 | 36.1 | 29.0 | 2.4 | 10.77 | .205 | .196 | .671 | 1.105 | 1.545 |
| 7500 | 368 | 270.3 | 344.3 | 68.8 | .785 | 37.0 | 29.3 | 2.8 | 10.73 | .203 | .222 | .766 | 1.232 | 1.642 |
| 8000 | 392 | 279.6 | 349.7 | 65.2 | .800 | 37.8 | 29.5 | 3.3 | 10.66 | .202 | .251 | .871 | 1.374 | 1.732 |
| 8500 | 417 | 289.2 | 354.5 | 60.7 | .816 | 38.5 | 29.7 | 3.8 | 10.58 | .200 | .289 | 1.010 | 1.561 | 1.831 |
| 9000 | 441 | 298.5 | 358.9 | 56.2 | .832 | 39.2 | 29.9 | 4.5 | 10.43 | .199 | .333 | 1.172 | 1.784 | 1.914 |
| 9500 | 466 | 307.7 | 362.9 | 51.3 | .848 | 39.8 | 30.1 | 5.2 | 10.28 | .198 | .388 | 1.371 | 2.058 | 1.994 |
| 10000 | 490 | 316.9 | 366.6 | 46.2 | .864 | 40.3 | 30.3 | 6.0 | 10.10 | .197 | .458 | 1.623 | 2.409 | 2.068 |
| 10500 | 515 | 326.0 | 370.2 | 41.1 | .881 | 40.8 | 30.4 | 6.8 | 9.95 | .196 | .543 | 1.938 | 2.843 | 2.142 |
| 11000 | 539 | 335.0 | 373.6 | 35.9 | .897 | 41.3 | 30.6 | 7.7 | 9.75 | .196 | .657 | 2.349 | 3.415 | 2.202 |
| 11500 | 564 | 344.0 | 376.5 | 30.2 | .914 | 41.8 | 30.7 | 8.7 | 9.56 | .195 | .821 | 2.949 | 4.252 | 2.263 |
| 12000 | 588 | 353.1 | 379.9 | 24.9 | .929 | 42.2 | 30.9 | 9.7 | 9.36 | .194 | 1.045 | 3.765 | 5.393 | 2.312 |
| 12500 | 613 | 362.0 | 382.6 | 19.2 | .946 | 42.6 | 31.0 | 10.8 | 9.15 | .194 | 1.424 | 5.150 | 7.331 | 2.361 |
| 13000 | 637 | 370.9 | 385.3 | 13.4 | .963 | 43.0 | 31.1 | 12.0 | 8.94 | .193 | 2.129 | 7.728 | 10.945 | 2.402 |
| 13500 | 662 | 379.8 | 388.0 | 7.6 | .979 | 43.4 | 31.2 | 13.2 | 8.74 | .192 | 3.898 | 14.204 | 20.024 | 2.440 |
| 14000 | 686 | 388.6 | 390.6 | 1.9 | .995 | 43.7 | 31.4 | 14.4 | 8.53 | .192 | 16.685 | 60.814 | 85.432 | 2.470 |
| 14280 | 700 | 389.9 | 389.9 | .0 | 1.000 | 43.8 | 31.4 | 14.8 | 8.44 | - | - | - | - | - |

* concerns with condensate freeze-up on cooling coil at air speed below 300 fpm, and saturated suction temperature below 30 F

Figure 1. Existing Cooling and Dehumidification System Schematic




SYSTEM PERFORMANCE

| | North Unit | South Unit |
|---|---|---|
| Airflow (cfm) | 14,615 | 12,372 |
| RA Temp. (F) | 74.6 | 72.7 |
| RA Rel.Hum. (%) | 42.2 | 42.9 |
| SA Temp. after Fan | 50.4 | 47.9 |
| SA Temp after Cooling Coil (F) | 49.7 | 47.3 |
| Fan Power Draw (kW) | 6.8 | 6.2 |
| Cond. Unit Power Draw (kW) | 37 | 36 |
| Total Cooling Capacity (tons) | 35.7 | 32.0 |
| Sens. Capacity (tons) | 32.6 | 28.1 |
| Sens. Heat Ratio | .914 | .879 |
| Deh. Rate (lb/hour) | 34.3 | 43.2 |

Figure 2. Psychrometric Process: Cooling and Dehumidification with Air Reheat

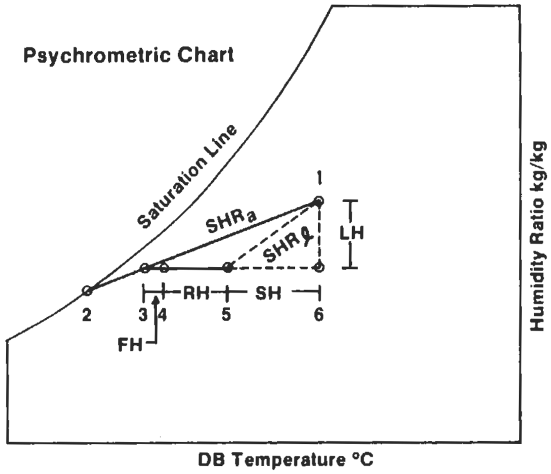



Figure 3. Cooling and Dehumidification Process SHR vs. Cooling Coil ADP Temperature

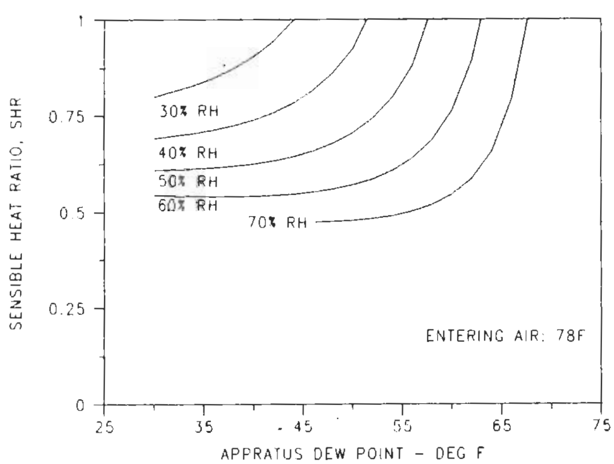



Figure 4. Dehumidification Capacity and Coil Face Velocity vs. Cooling Coil SST.

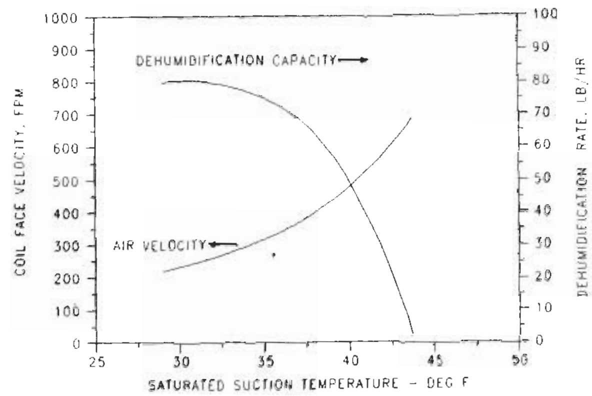



Figure 5. SHR and System EER vs. Cooling Coil SST.

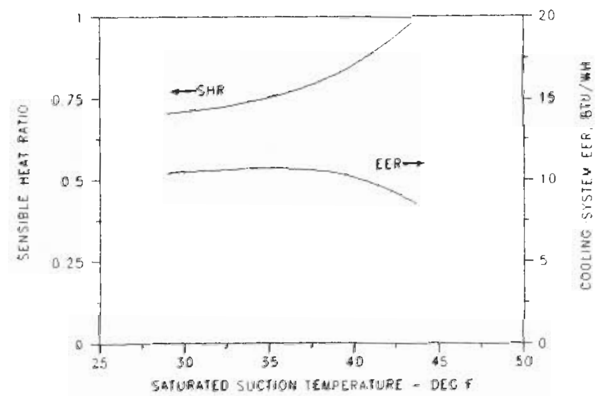



Figure 6. Dehumidification and Cooling Energy Requirement vs. Cooling Coil SST for Load SHR of 0.55.